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Vibration failure in admission pipe of a steam turbine due to flow instability

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ABSTRACT

High vibrations in admission piping of a steam turbine were analyzed. Vibration failure was detected after piping modification as part of upgrading a 300 MW power turbine plant searching for 10% power increment. However, after 1 year operation a vibration malfunction was detected in control valve and fittings of income piping with risk of cracking for maximum output. A study of computational fluid dynamics (CFD) revealed large steam flow instabilities produced by recirculation and high velocity exceeding a critical point. Measurements of natural frequency piping system with the turbine stall and subsequent measurements of frequency and vibration analysis during turbine operation indicated that recirculating flow plays a main role in the vibration problem by resonance. The paper discusses CFD results obtained with a proposed pipe configuration that reduces turbulence effects. Combined pressure slide and diameter increment in piping lead to reduced vibration turbine operation.

1. Introduction

Single and multi-phase flows may produce pipe vibration due to unstable velocity components, especially in turbulent regime or because of unsteady flow behavior like oscillating and pulsating flows. Vibration occurs since induced forces meet specific conditions. If the frequency of induced forces corresponds with pipe geometry's first or second modes of frequency, the conditions are favorable for setting up resonance. Vibration may produce small and large stresses on pipe materials and support system which can lead to failure. Recent studies [1] have been conducted to measure and calculate time-dependent forces in T-junction piping with the objective of defining a relationship between the characteristics of forces and the parameters of flow and pipe geometry, in order to predict appearance of vibration and its effects. Ibrahim [2] reviewed the literature, surveying the subject of mechanics of pipes by problem arising from conveying fluid, such as fluid-elastic instability under conditions of turbulence.

Furthermore, Sarpkaya [3] conducted a review of vortex induced vibration in circular cylindrical structures under steady uniform flow. From these reviews it is concluded that researchers currently use modeling, dynamic analysis and stability regimes to characterize and interpret vibration effects. Also, efforts have been done for obtaining exact solutions from data points obtained *in situ* though restrictions of applicability prevent their use as a generalized method of diagnostics [4]. Others use combined methods of failure analysis starting by visual inspection during different stages of plant processes operation involving piping systems, like reported by Porter et al. [5]. After a first insight into the problem, which involves realizing that piping and piping supports were properly designed, the attention focuses on the measurement of frequency of vibration in order to disregard resonance. A usual practice based on valve path opening control follows [6]. For instance,

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flow pulsation can be apparently controlled by adjusting valve path opening levels, but it commonly leads to low load rate operation in what it is considered a temporal solution only, although for some cases this procedure has represented a definitive solution [5].

Jiang et al. [7] used computational flow dynamics (CFD) to analyze combined fluid–structure vibrations in a centrifugal pump by resolving the fluid unsteadiness and using data transfer from each other domains; vibration modes at blade passing frequencies were extracted too and used for noise reduction. However, the topic is still open because there are evidences that fittings, included into the piping-valve system upstream–downstream the valves like sphere elbows, may also represent a source of flow unsteadiness [6].

The present study represents a combined method of failure analysis including visual observation, frequency measurements and CFD analysis of steam flow in a T-junction, which is located downstream the governor valve and before steam turbine inlet [8,9]. Results from CFD were used for completing the failure analysis by proposing a new piping configuration that represented lower steam velocity at turbine inlet and reduced flow recirculating in exit branch of T-junction. A final solution to the high vibration problem involved the new piping configuration and a modified path valve opening sequence.

Fig. 1. General view of refurbished HP and IP rotor.

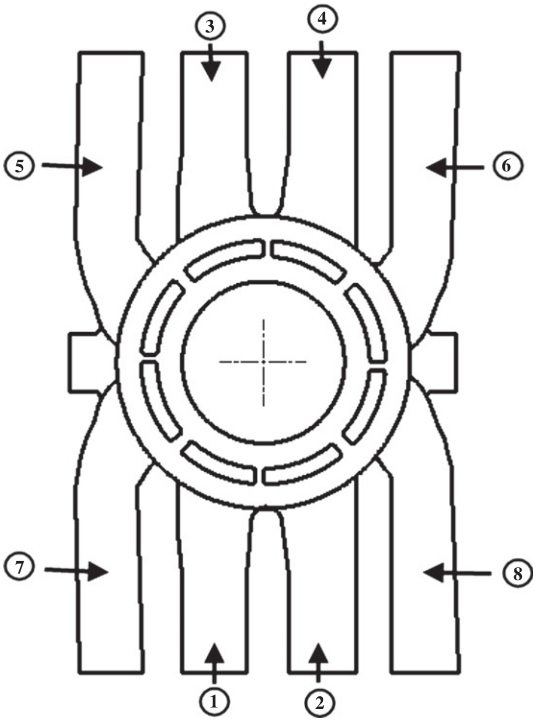

Fig. 2. Original configuration of nozzle groups.

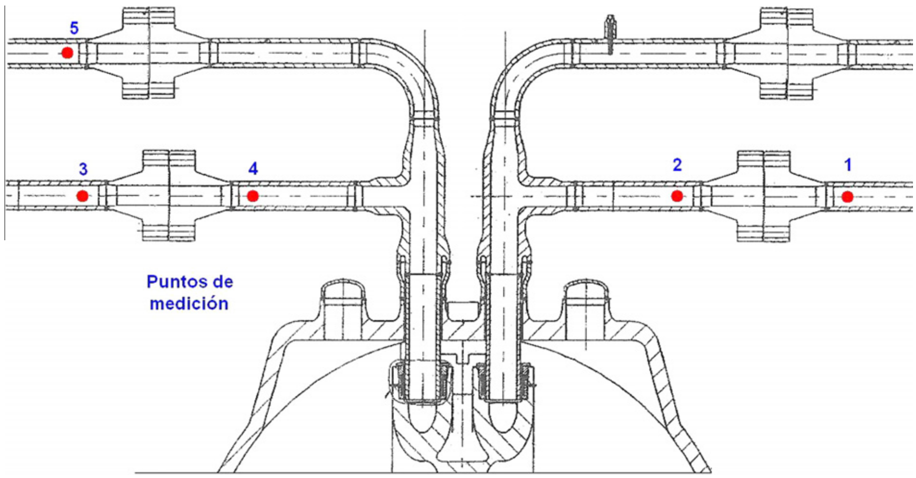

Fig. 3. Schematic representation of upper spaghetti after modification.

After 1 year of successful operation of upgraded steam turbine from 300 to 330 MW a faulty operation of the control valve No. 2 was detected. It is worth to mention that new rotors (new steam turbine path) were installed as a part of upgrading process, which are shown in Fig. 1.

Also the eight original control valves were substituted with new pipe distribution system consisting in six control valves as shown in Fig. 2. Originally, there was an eight nozzle group fed from each control valve by individual admission pipe (spaghetti).

Control valves operated sequentially. After modification all six valves fed steam to one steam chamber prior to first stage turbine because control stage was eliminated. In this case the control valves were operated simultaneously. When faulty operation was detected the turbine was shut down to correct the malfunction. Then, when increasing load to 280 MW very strong vibrations in one upper admission pipes were detected. Vibrations increased to 80 mm/s, so it was not possible to raise turbine load any further. At this stage an investigation was started to clear out the causes of the high vibrations. After preliminary investigations a sequence of activities to find out the causes of the fault was conducted as follows:

(a) vibrational analysis (spectrum) of admission pipe (spaghetti) at power rates below and above 280 MW, where strong vibrations were detected;
(b) revision of the operational data during event;
(c) measurement of natural frequencies of spaghetti;
(d) to get all dimensions and geometry of admission piping system;
(e) building up a CFD model of steam flow through piping system;
(f) conducting turbine operational tests like increasing load rate and to starting-up under sliding pressure procedure (control valves fully opened).

Piping spaghetti configurations for before and after modifications are shown in Figs. 2 and 3 respectively. As it is observed, one portion of modifications included four upper control valves, which were substituted by two control valves of different size, being new ones of larger dimensions than previous.

2. Vibrations measurement

2.1. Measurement of vibrations during load variation, around 280 MW

The location of point 1 where vibration was measured is indicated in Fig. 4 in agreement with Fig. 3. Results of vibration measurements in steam flow piping are given in gravities, g_n, in Figs. 5 and 6, for power rates below and above 280 MW, respectively. It is observed that vibration pipe amplitude is lower in power rates below 280 MW. Therefore, by comparing the vibration at low frequency for power rates below and above 280 MW a peak of $1.1g_n$ is observed at 14.64 Hz as shown in Figs. 5 and 6.

2.2. Measurement of natural frequency of admission pipes (spaghetti)

Natural frequency of piping system here called spaghetti was measured after turbine shut down. Pipe was excited using a load cell hammer and the response signal was processed employing an accelerometer in order to obtain its spectrum, which is presented in Fig. 7.

Fig. 4. Pipe for turbine steam feeding system indicating location for vibration measurement after piping modification.

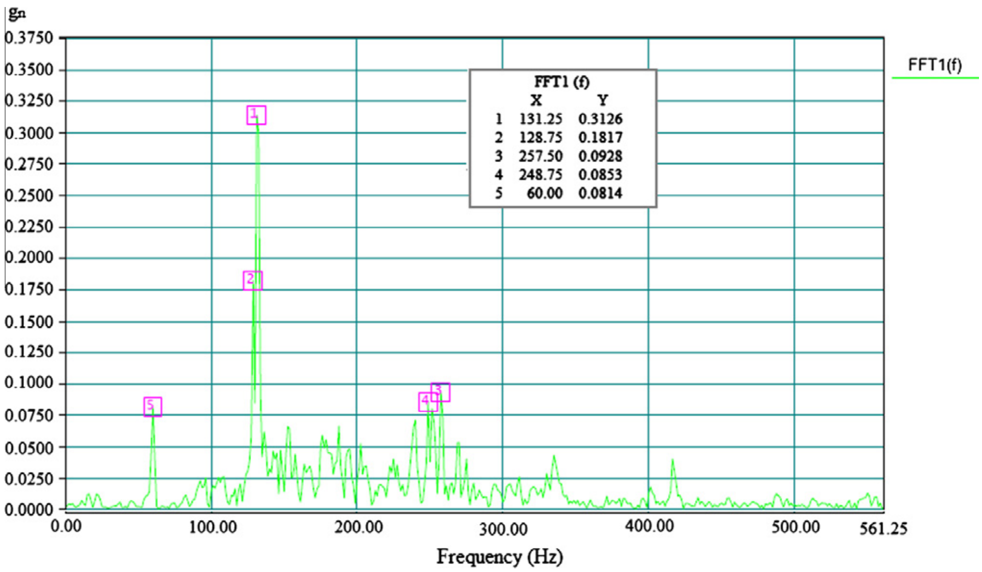

Fig. 5. Vibration spectrum of upper spaghetti below 280 MW load. Scale of V in mm s^{-1}.

As it is observed in Fig. 7, the first natural frequency of spaghetti corresponds to 14.06 Hz [9].

3. Analysis of operational parameters at 284 MW load

Measurements of pressure and flow rate were conducted while turbine engine was operating on refurbished piping configuration. Steam pressure oscillations were detected in a gauge located upstream the turbine, since there is no control stage, for turbine load in agreement with spaghettis strong vibration beginning.

Pressure amplitudes up to 4 bar were registered as it is observed in Fig. 8. Further, steam flow oscillation was detected as it is shown in Fig. 9. Steam flow rate oscillates in a quasi periodic path, which makes it easy to identify similar pattern in other signals as discussed below.

Steam pressure and flow rate oscillations and other parameters showed temporal instability as well, like output power, which is shown in Fig. 10. Another source of instability was found by analyzing the vibrations pattern of bearings. This last oscillated but increments remained under acceptable margin.

Research investigations were based on a combination of data from measurements and operating conditions, which lead to establish a former hypothesis that steam flow rate oscillations first occur, and consequently steam pressure downstream, which excited spaghetti natural frequency.

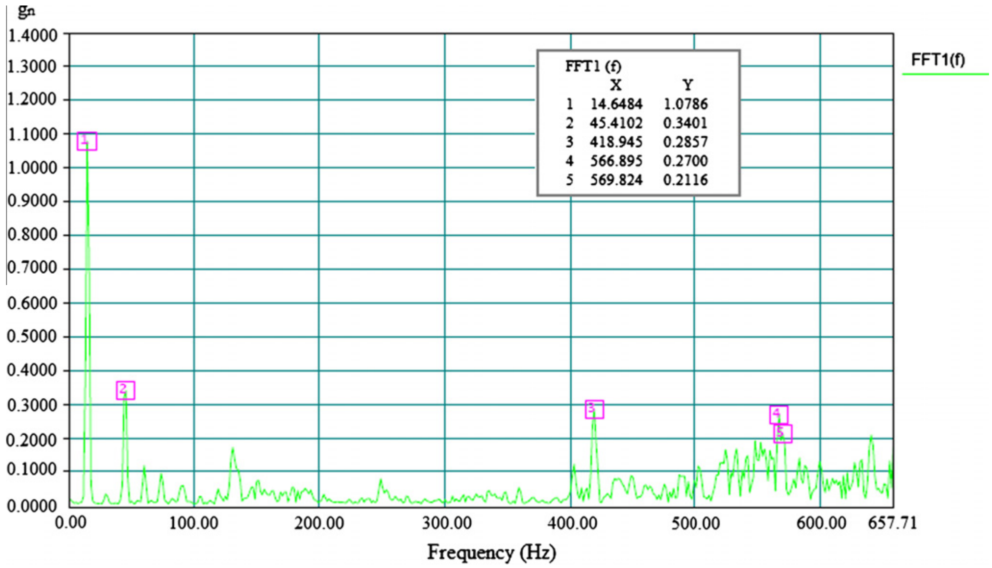

Fig. 6. Vibration spectrum of upper spaghetti at 284 MW load. Scale of V in mm s^{-1}.

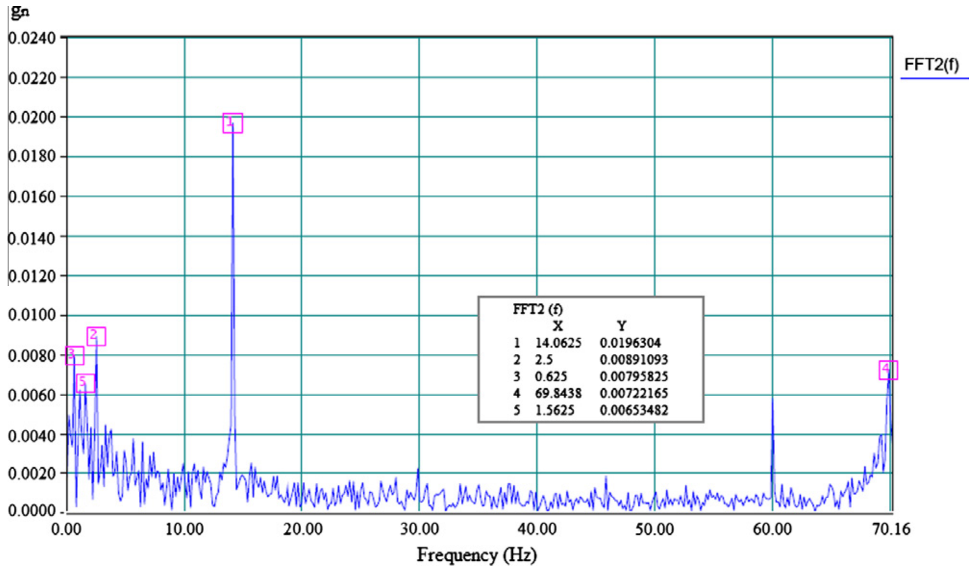

Fig. 7. Spectrum of frequencies measured *in situ*. Scale of g_n in m s^{-2}.

This caused a piping vibration with very large amplitude representing a huge risk of fracture in the piping system. Incidentally, cracks were detected in the spaghetti, which occurred presumably due to vibrations. This topic was not investigated in deep, and it is left for further report.

Therefore, steam flow rate instability was suspected as the main cause of oscillations observed through spaghetti vibration. Afterwards, it was decided to conduct a numerical analysis based on CFD, by constructing a model of the flow in the admission piping, in order to predict flow behavior that may lead to clarify what kind of instability may have an effect into the faulty process at hand.

4. Computational modelling of flow in spaghetti

The steam flow in a T pipe junction was analyzed with a CFD computer program Fluent [10]. All pipe dimensions were obtained *in situ*, as well as geometry details. From these, the computational model was constructed with a combination of non-structured/structured mesh elements [11]. The computational representation of one piping connecting to the turbine was developed and used to analyze flow behavior. CFD piping system is shown in Fig. 11.

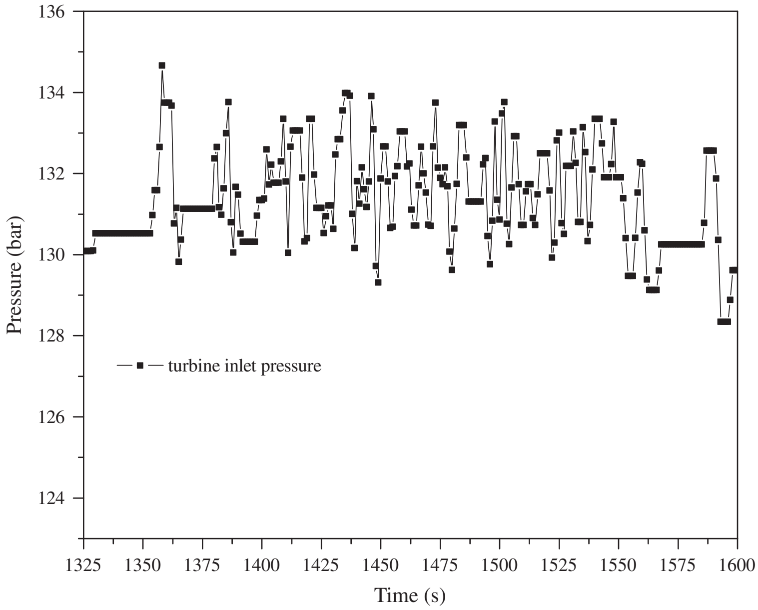

Fig. 8. Steam pressure temporal behavior in turbine pipe feeding.

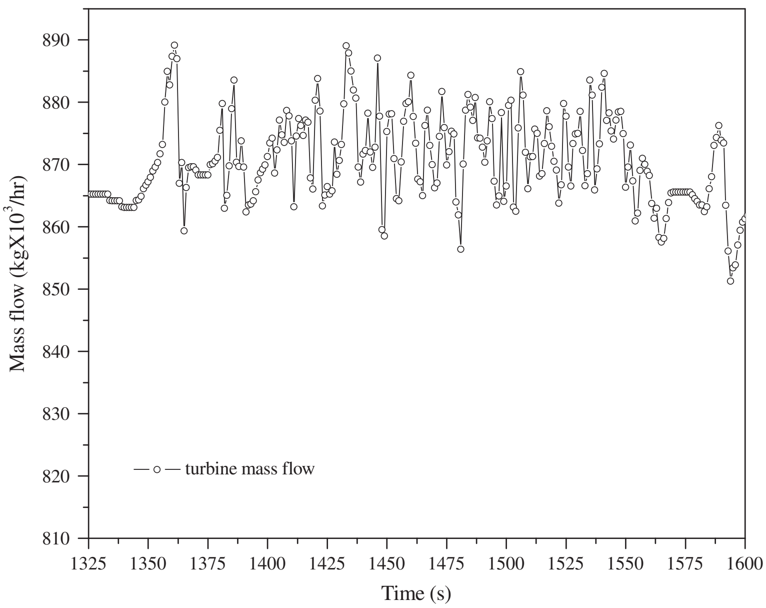

Fig. 9. Steam flow oscillation during event at 284 MW.

A computation of flow conditions was obtained. The solution was based on the use of a Reynolds stress model, RSM, for modeling turbulence and boundary conditions defined on flow rates sown in Fig. 11. A critical velocity was calculated from an expression for velocity of sound assuming sound wave as a small pressure disturbance and considering steam as an ideal gas at turbine operating conditions where sound wave displaces [12]:

$$c_a = \sqrt{kRT} \tag{1}$$

where k is a specific heat ratio, R is the gas universal constant, and T is steam temperature given in K. A sonic velocity in steam, c_a = 697.6 m/s was calculated in pipe near turbine. Further, CFD computations served to identifying a zone of recirculating flow in branch T-junction, characterized by vortex structure, which is shown in Fig. 12a and b by contours and vectors of velocity, respectively.

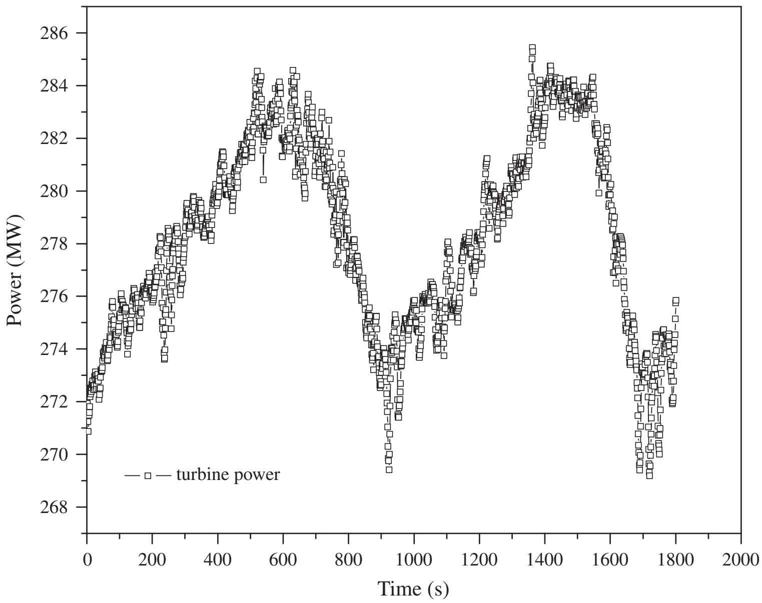

Fig. 10. Oscillation of output power rate under unit operation with modified piping system.

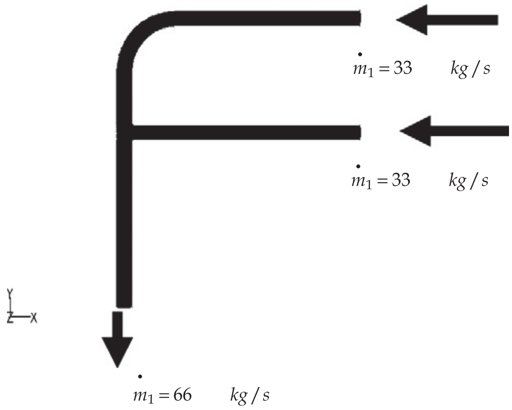

Fig. 11. Schematic representation of the piping connection in the CFD solution.

The numerical simulation included a right angle piping connection following a real dimensions T-junction. It must be emphasized that recirculation has also been observed in T-junctions with chamfer transition between branch and main-stream pipe [13]. However, the use of smooth curve transition may help by reducing recirculation strength by 10–15%. Chamfered junctions produce a separating flow region that allows for boundary layer development in a length downstream smaller in diameters compared to the case of right angle T-junction [13].

It has been documented that an effect of this flow cell is that it reduces an effective area for flow within the pipe, leading to strong velocity gradients, since mass conserve and pressure imbalance is local [14].

The flow accelerates out of recirculation flow region several diameters downstream to the T-junction. Increased velocity in this case reached more than 200 m/s. Furthermore, recirculation cell flow after the T-junction is basically unstable and characterized by complex dynamics, with random variations in time and space, which may produce big oscillations of pressure that may travel ahead and backwards [15,16]. However, no evidences have been documented in order to quantify vibrations produced by flow instabilities in T-junctions alone. Literature published works in this topic are rather scarce and no definitive conclusion can be made on fluid–structure interaction induced vibration.

A new piping system configuration was proposed, which included a gradual increment of pipe diameter that connects with turbine, as observed in Fig. 13.

The flow in new piping was simulated with the same boundary conditions as original one. The results are shown in contours maps and vectors of velocity in Fig. 14a and b, respectively. A recirculating flow region is still observed downstream the T-junction. However, magnitude of velocity in around recirculation flow cell and the opposite wall to the connected pipe

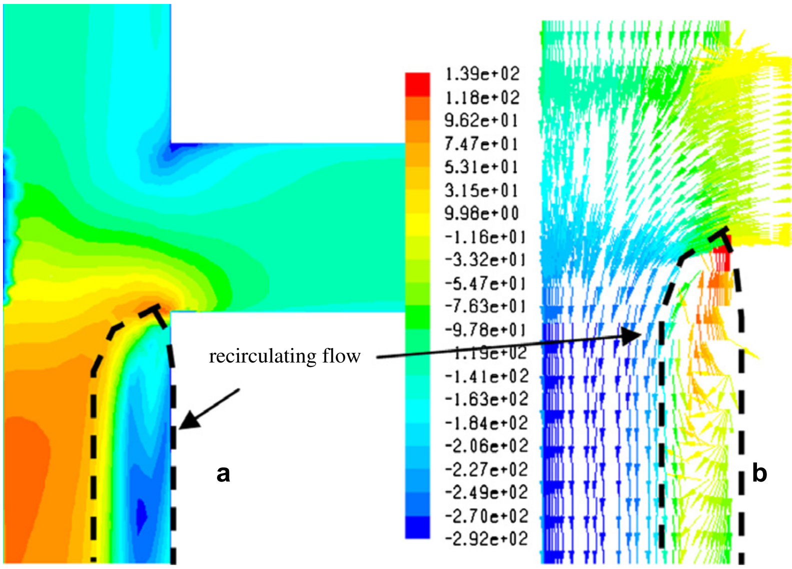

Fig. 12. The flow structure in the region of the T-junction of the spaghetti; (a) contours of velocity and (b) vectors of velocity.

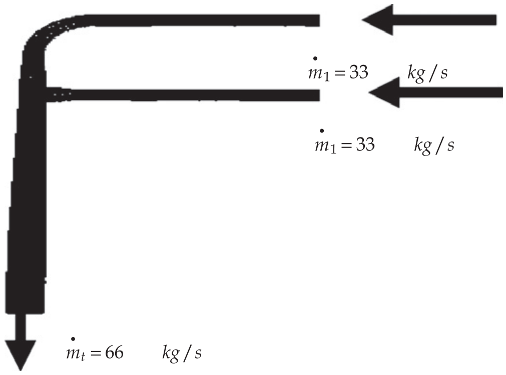

Fig. 13. Schematic representation of the spaghetti: CFD proposed solution.

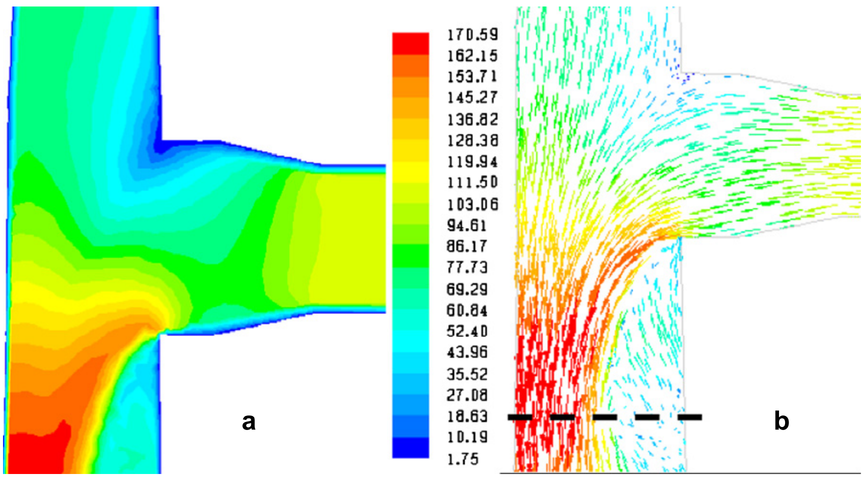

Fig. 14. The flow structure in the region of the T-junction of the spaghetti for the proposed configuration; (a) contours of velocity and (b) vectors of velocity.

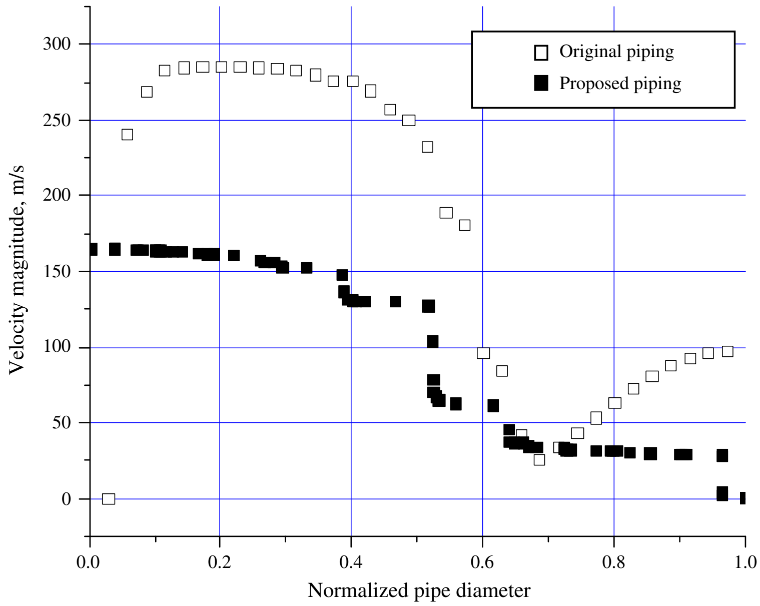

Fig. 15. Velocity profiles from CFD simulation downstream T-junction according to Fig. 14.

drop down from 292 to 160 m/s, rendering less instability and impacting on Mach number. A velocity profile was built downstream the T-junction, which is shown in Fig. 15 for original and proposed piping systems. Magnitude of velocity drop makes the proposed piping suitable to reducing risks of resonance associated with big oscillations of spaghetti piping system. It is envisaged that proposed piping will produce less vibration to the turbine feeding system in future. Also, studies of turbulence intensity and other turbulence parameters are recommended in order to characterize their frequency and its connection to natural frequencies of piping.

5. Analysis of results

Over the period of increasing turbine load above 280 MW, extremely high vibrations of upper spaghetti were recorded. Higher values vibrations reached 80 mm/s at 14.64 Hz frequency. Steam pressure oscillation upstream turbine first stage was

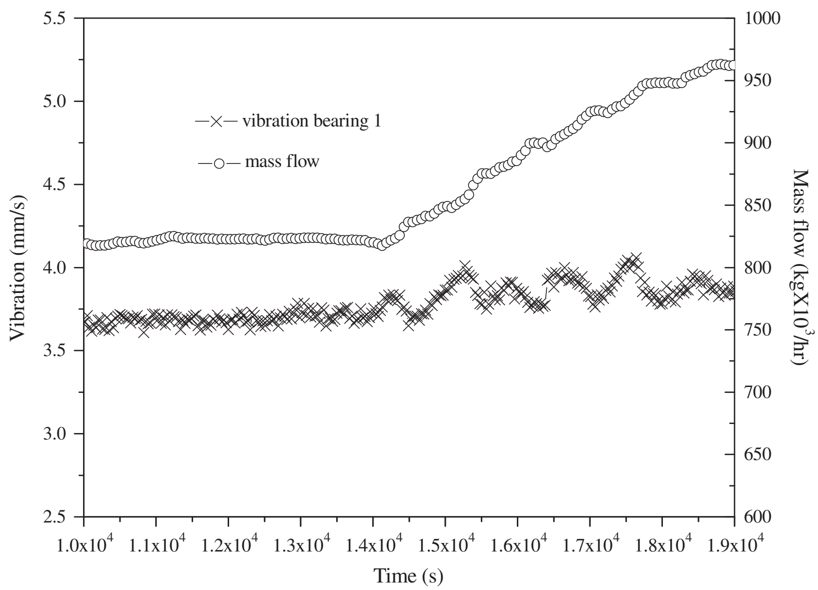

Fig. 16. Real time monitoring of turbine parameters.

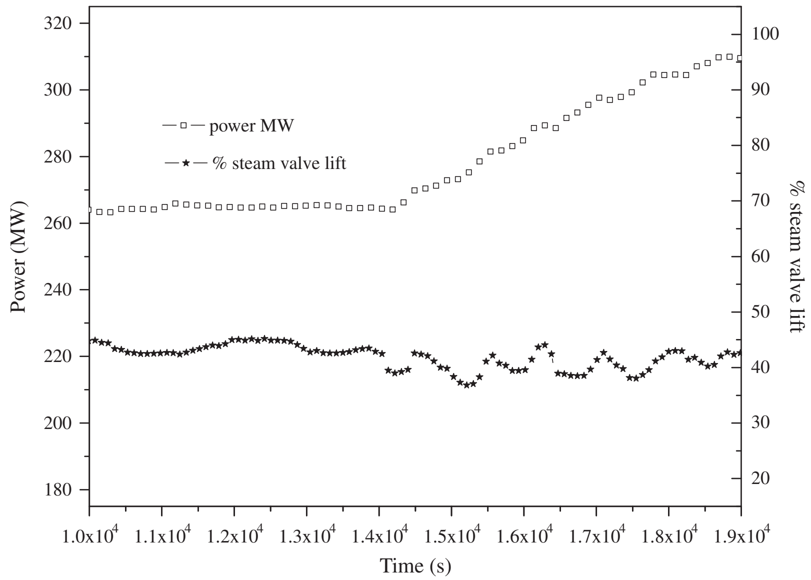

Fig. 17. Monitoring image of the turbine parameters, with the control valve position fully opened.

recorded. The amplitude of this oscillation was about 4 bar with a frequency close to 14.6 Hz. This oscillation excited the first natural frequency of spaghetti, which was about 14.06 Hz, measured under conditions of non-operating turbine and pipe material at low temperature. The first analysis leads to define this as proper scenario for resonance phenomenon appearance.

On the other hand, CFD analysis indicated that recirculation flow at the T-junction effectively diminishes the cross section area for flow, especially for pressure differences registered upstream and downstream the control valves; leading to increasing steam flow velocity. This second analysis reinforced the assumption that the application of a T-junction in the steam admission piping upstream is one cause of flow instabilities.

Furthermore, reducing the number of control valves installing two larger valves on top of high pressure HP turbine caused admission steam unbalance arriving to first stage.

Based on the last step of analysis, it was recommended to start up the turbine in the sliding pressure mode, revising control valves and control system.

6. Turbine operation test

After conducting a careful inspection of control system, the turbine was started up. The load was raised from 200 to 300 MW with control valves opened between 35% and 45%. No oscillation of pressure before upstream the turbine neither high vibrations of spaghetti occurred. The test was repeated three times with positive results.

After that it was decided to change the load with different load increment ratios. When the load was changed with increments of 3–4 MW/min no oscillations occurred. The results are shown in Fig. 16, where it is observed that mass flow rate increases smoothly for power rates below and above 280 MW, while vibration remains below 4 mm s^{-1}.

The signal for turbine power rate, shown in Fig. 17, confirms that governor valve lift did not overpass from 45%.

7. Conclusions

Steam pressure oscillations were detected leading to analyzing the piping system used to feeding a steam turbine. It was concluded that modification to piping induced vibrations to the turbine, and they were provoked by flow instability in a T-junction combined with specific conditions of flow and pressure upstream and downstream control valves.

It was deduced from registered operating data that oscillation frequency coincided with natural frequency of refurbished piping, which created resonance condition producing very high spaghetti (pipe) vibrations.

Even small errors in the control valve system and unbalanced flow to the turbine admission chest may contribute to induce this kind of flow instability.

It was concluded that the use of a T-junction connector is not adequate in this case.

A proposed temporary solution based on sliding pressure mode (with control valves opened) and with load increments between 3 and 4 MW/min permitted to avoid the flow and pressure oscillation (instabilities). Also fine control of valve was tested with good results, responded positively and starting up the turbine.

8. Recommendations

Finally it was recommended to study and test other alternatives of the safe load changes of the turbine. It was proposed to change adjustments of the control valves thus that the upper valves will start to open about 3–5% of full stroke before other four valves commence to open.

Other recommendations included to design new starting up curves and realize thermodynamic calculation of the modified turbine using stage by stage computer program. It would be quite costly to return back to the original design of eight valves before exploring other possibilities.

Acknowledgement

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References

[1] Riverin JL, de Langre E, Pettigrew MJ. Fluctuating forces caused by internal two-phase flow on bends and tees. J Sound Vib 2006;298(4–5):1088–98.
[2] Ibrahim RA. Mechanics of pipes conveying fluids – Part II. Applications and fluidelastic problems. J Press Vess Technol 2011;133:024001.
[3] Sarpkaya. A critical review of the intrinsic nature of vortex-induced vibrations. J Fluids Struct 2004;19(4):389–447.
[4] Saha S. Estimation piping of point vibration loads from field measurements for industrial piping. J Press Vess Technol 2009;131:031206.
[5] Porter MA, Martens D, Harrylal R, Henley Ch. Valve-induced piping vibration. In: Proc conf ASME pressure vessels and piping. Paper PVP2011-57391, 2011; p. 899–904.
[6] Ziada S, Sperling H, Fisker H. Vibration of a high-pressure piping system due to flow in a spherical elbow. J Fluids Struct 2001;15(5):751–67.
[7] Jiang YY, Yoshimura S, Imai R, Katsura H, Yoshida T, Kato C. Quantitative evaluation of flow-induced structural vibration and noise in turbomachinery by full-scale weakly coupled simulation. J Fluids Struct 2007;23:4.
[8] Kubiak J, Sierra-Espinosa FZ. Results of the suggested tests to the steam turbine no. 4 in the thermal station Tula and recommendations to avoid vibrations and flow oscillation in the pipes spaguetti. Final Report, Universidad de Guanajuato, November 2008 [in Spanish].
[9] Kubiak J, Estrada D, Rodriguez JA. Natural frequency analysis in pipes spaguetti 7 and 8 of the steam turbine no. 4 of Tula thermal station Francisco Pérez Ríos Partial Report 3, Universidad de Guanajuato, July 2008.
[10] Fluent V6.3 User's Guide, Fluent Inc. Canterra Resource Park, vol. 3, 10 Cavendish Court, Lebanon NH 03766, 2001.
[11] Gatski TB, Hussaini MY, Lumley JL. Simulation and modeling of turbulent flows. NY: Oxford University Press; 1996.
[12] Van Wylen GJ, Sonntag RE. Fundamentals of classical thermodynamics. 2nd ed. New York: Wiley Inc.; 1973.
[13] Sierra-Espinosa FZ, Bates CJ, O'Doherty T. Turbulent flow in a 90 degree pipe junction: Part 1. Decay of turbulence fluctuations as the flow approaches the branch exit. Comput Fluids 2000;29:197–213.
[14] Sierra FZ, Adamkowski A, Urquiza G, Kubiak J, Lara H, Janicki W. The effect of boundary layer separation on accuracy of flow measurement using the Gibson method. In: Proc ASME conf on fluids engineering. Paper FEDSM07-37564, San Diego CA, August 2007.
[15] Paál G, Pinho FT, Maia R. Numerical predictions of turbulent flow in a 90° tee junction. In: 12th Int conf on fluid flow technologies, Budapest, Hungary, September 3–6, 2003.
[16] Pérez-García J, Sanmiguel-Rojas E, Viedma A. New experimental correlations to characterize compressible flow losses at 90-degree T-junctions. Exp Therm Fluid Sci 2009;33:261–6.